Industrial applications for LIGA-fabricated micro heat exchangers

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ABSTRACT

One of the well-known benefits of micro scale is enhanced heat transfer. This fact provides the motivation for fabricating a variety of micro heat exchangers using derivatives of the LIGA micromachining process. These heat exchangers can be made of polymers (PMMA), nickel (electroplated or electroless), or ceramics (Si_3N_4 and alumina are presently being investigated). These heat exchangers are envisioned for applications such as gas turbine blades, mechanical seals and/or bearings, boilers, condensers, radiators, evaporators, electronic component cooling, and catalytic converters. In this paper, methods to fabricate an array of heat exchangers for different applications are described. In addition, simple analytic models that illustrate the motivation for fabricating micro cross flow heat exchangers are shown to compare favorably with experimental heat transfer results.

Keywords: LIGA, cross flow heat exchanger, mechanical seal, silicon nitride HARMs,

INTRODUCTION

Derivatives of the LIGA process[1,2,3,4,5] are being used to fabricate a variety of components for use in heat transfer or mass transfer applications which include gas turbine component cooling, mechanical seal cooling[6], and applications where efficient, lightweight cross flow heat exchangers are desired[7], and efficient catalytic converters. To date, heat exchangers have been fabricated from either polymers or metal, and the feasibility to produce Si_3N_4 components with LIGA-like features has recently been demonstrated. Devices that have been fabricated to date are described as follows:

1) A cross flow micro heat exchanger has been fabricated from PMMA using derivatives of the traditional LIGA process[7,8]. A cross flow heat exchanger is used in a wide variety of applications, usually between a gas (which passes through the plane of the heat exchanger), and a liquid (which is contained within the plane of the heat exchanger). A cross flow heat exchanger is used in cases where the pressure drop of at least one fluid (usually the gas) needs to be minimized

2) Another set of applications involves mechanical seals[6,9]. Many mechanical seal failures are induced by thermal distortion resulting from heat generated by friction at the seal interface. A prototype seal has been fabricated using yet another derivative of the LIGA process in which a micro heat exchanger is fabricated just beneath the load-bearing surface. This electroplated nickel heat exchanger very effectively removes heat from the seal interface and prevents thermal distortion-related failures. Interestingly, recent manufacturing advances will soon make it possible to fabricate such a product from a high performance ceramic, Si_3N_4.

3) Gas turbine component cooling is yet another area that utilizes micro heat exchangers[10]. Methods are being developed to electroplate micro heat exchangers onto curved surfaces. Pin-fin micro heat exchangers can be used to more effectively cool gas turbine components such as stators and blades, thereby increasing the efficiency of the gas turbine. Efforts are underway to electroplate such pin-fin heat exchangers from alloys with excellent high temperature properties, or to cast these micro heat exchangers from Si_3N_4.

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4) Finally, micro catalytic converters, with dimensions similar to those shown in Figure 2 are currently being fabricated. The motivation for a micro catalytic converter is similar to that of a heat exchanger: more reaction/unit volume as a result of the length scale of the channels.

The harsh environments (high friction, high temperature, high stress) in which many of these micro heat exchangers/catalytic converters will operate necessitate materials with superior properties. One example of efforts to address this issue is a recent collaborative effort with Honeywell Ceramics, Inc. Recent efforts have generated great optimism that at parts with LIGA-like structures can be manufactured for some of the applications from Si_3N_4 using a "lost mold" process (the mechanical seal, certain gas turbine components, the catalytic converter). By filling a polymer mold such as shown in Figure 1b with a ceramic precursor, fully dense silicon nitride structures have been formed. The final part has the fine features characteristic of the LIGA process, coupled with the excellent high temperature, wear resistant properties of Si_3N_4.

This paper will describe manufacturing methods used to fabricate micro heat exchangers for different applications from a variety of materials. In addition, in the case of the cross flow heat exchangers, experimental data is shown to compare favorably with simple analytic model predictions.

FABRICATION AND TESTING

1) Micro cross flow heat exchangers

To fabricate a polymer cross flow micro cross flow heat exchanger, a nickel mold insert (Figure 1a) is fabricated using the traditional LIGA process. Two polymer halves (Figure 1b) are molded and then bonded together to produce the product shown in Figure 1c.

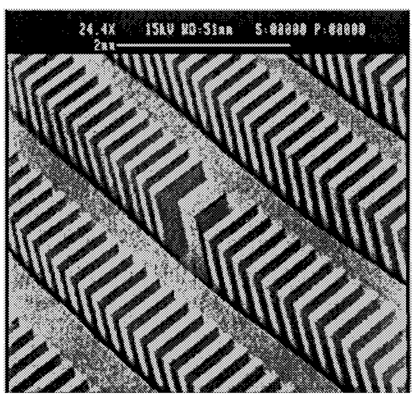

Figure 1a: Nickel mold insert

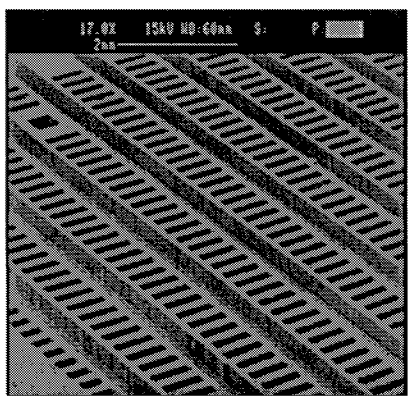

Figure 1b: PMMA embossed halves

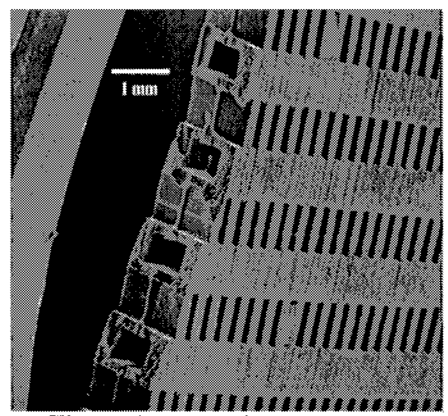

Figure 1c: Cross flow heat exchanger after bonding together two halves

A derivative of the LIGA process is used to fabricate another version of the cross flow heat exchanger, this time from nickel. This heat exchanger has greater strength, greater performance, and can operate at a higher temperature., Again, a mold insert (Figure 2a) is used to mold polymer parts perforated with holes of hexagonal cross section (Figure 2b). An electroless plating process is used to coat the entire surface of the polymer (except two opposing edges) with nickel. After electroless plating, the part is immersed in a solvent to dissolve the polymer. The resulting heat exchanger is shown in Figures 2c- 2e.

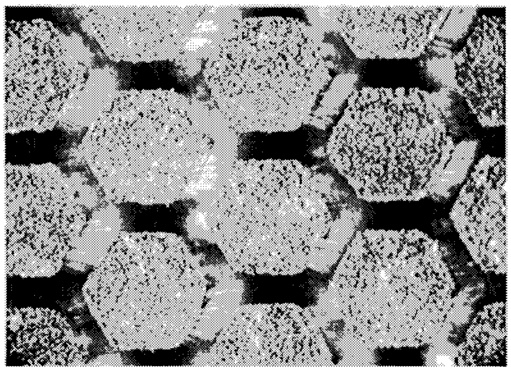
Figure 2a: Nickel mold insert (height of posts 1 mm, length of side of hexagon= 250 μm, gap between parallel sides= 150 μm

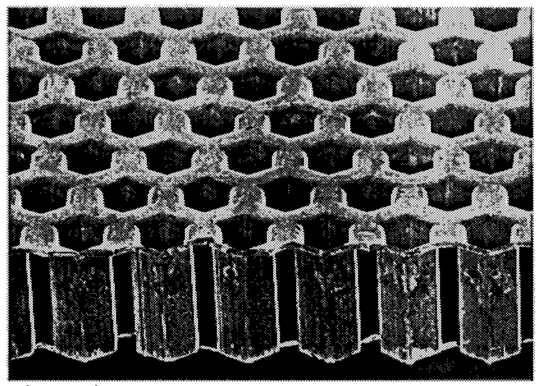
Figure 2c: Nickel heat exchanger after electroless plating and removal of polymer

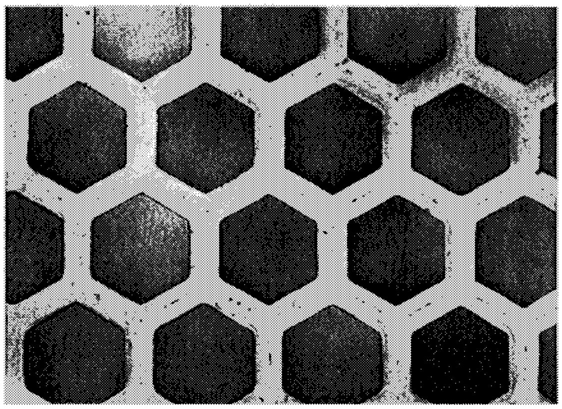
Figure 2b: Molded polymer sheet

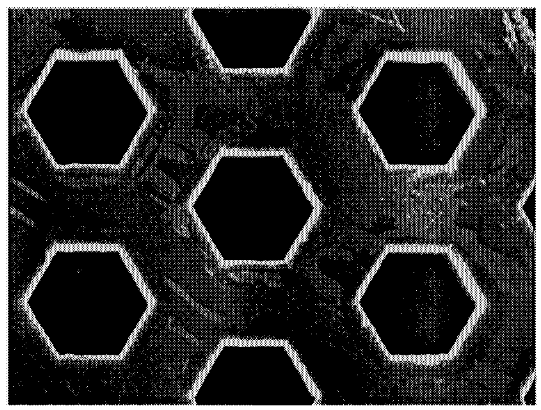
Figure 2d: View of nickel heat exchanger showing channels through plane of heat exchanger

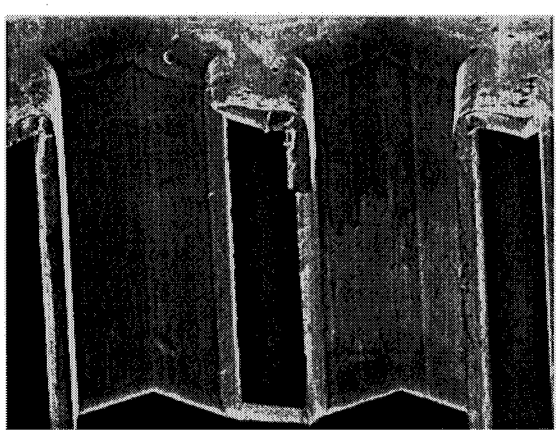
Figure 2e: View of cross section of in-plane channel

Both the polymer and nickel cross flow micro heat exchangers have been tested and the results have been compared to model predictions. For both heat exchangers, water and air were the two fluids exchanging heat, with water flowing through channels in the plane of the heat exchanger and air flowing through channels normal to the plane of the heat exchanger. Analytic models to predict the performance of both heat exchangers have been built. The model for the polymer heat exchanger is more complicated and is explained in detail elsewhere[7,8]. The slightly simplified forms of the equations used to predict the performance of the metal cross flow heat exchanger are provided by Equations (1) and (2). Equation (1) can be used to approximate specific heat transfer to the fluid (air) passing in cross flow through heat exchanger. The assumption has been made that the wall of the channel is effectively the temperature of the water, T_{water}, flowing in the plane of the heat exchanger, and the convection coefficient is constant along the length (true for fully developed flow, which is not the case). The rate of heat transfer to the gas/channel is given by Equation (2):

$$\frac{T_{water} - T_{exit}}{T_{water} - T_{inlet}} = exp\left[-\frac{C_1 k_f L}{\rho C_p V D_h^2}\right] \qquad \text{Equation (1)}$$

where:

C_1	=	A constant depending upon the Reynolds number,
k_f	=	the thermal conductivity of the fluid,
L	=	the length of the air channel (the thickness of the heat exchanger),
V	=	the velocity of the air in the heat exchanger,
D_h	=	the hydraulic diameter of the air channel,
C_p	=	the constant pressure specific heat of the air,
ρ	=	the density of the air,
T_{exit}	=	the exit temperature of the air from the heat exchanger, consistency of symbols
T_{inlet}	=	the temperature of the air entering the heat exchanger.

$$Q = \dot{m} C_p (T_{exit} - T_{inlet}) \qquad \text{Equation (2)}$$

where:

Q	=	The rate of heat transfer/channel to the gas (W)
$\dot{m}$	=	The mass flow rate of gas through the channel

Equation (3) provides the pressure drop of the air across the heat exchanger for the case where flow in the channel is fully developed and laminar. The first term on the right hand side accounts for viscous drag within the channel and often represents the dominant term. The second term accounts for inlet and exit losses associated with sudden expansion/contraction.

$$\Delta P = \frac{32 V L \mu}{D_h^2} + \frac{1.5 \rho V^2}{2} \qquad \text{Equation (3)}$$

where:

V	=	the velocity of the fluid through the channel, and
μ	=	the viscosity of the fluid.

Together, Equations (1) and (3) provide the motivation for fabricating micro cross flow heat exchangers. For given fluid material properties and a given velocity, the exit temperature of the gas flowing through the heat exchanger is constant for a given value of L/D_h^2. By decreasing the hydraulic diameter of the gas channels in the heat exchanger, the required gas channel length (heat exchanger thickness) decreases dramatically. *Surprisingly, the penalty normally associated with pumping fluid through micro channels is minimal in a cross flow micro heat exchanger* since, for a given velocity (assuming the inlet and exit losses are small relative to viscous drag losses), the pressure drop is proportional to the ratio of L/D_h^2. Therefore, to first order, for a given L/D_h^2, heat transfer/unit volume is inversely proportional to length while the pressure drop does not vary with scale.

For the case of the metal heat exchanger shown in Figures 2c-2e, experimental results (data points) at room temperature are compared to model predictions (solid lines) in Figures 3a and 3b[8]. In Figure 3a, the measured pressure drop of the air across the heat exchanger is plotted as a function of velocity (mass flow rate) compares favorably with the model prediction using a slightly modified form of Equation (2). In Figure 3b, the percent of the maximum possible heat transfer to the air is plotted versus velocity of the air through the channel. The flow rate of water was sufficiently high such that the difference in temperature between the water exiting and entering the heat exchanger was very small. This greatly simplifies the analysis, since the temperature of the walls of all the air channels can be assumed to have the same, uniform value. The maximum possible heat transfer is proportional to the difference in temperature between the air entering the channel and the temperature of the water (T_{water}-T_{inlet}) and the actual heat transfer is proportional to the change in temperature of the air passing through the heat exchanger (T_{exit}-T_{inlet}). Also shown are model predictions based on a slightly modified version of Equation (1) that does account for non-fully developed flow. Model and experimental results for the polymer heat exchanger also compared favorably.

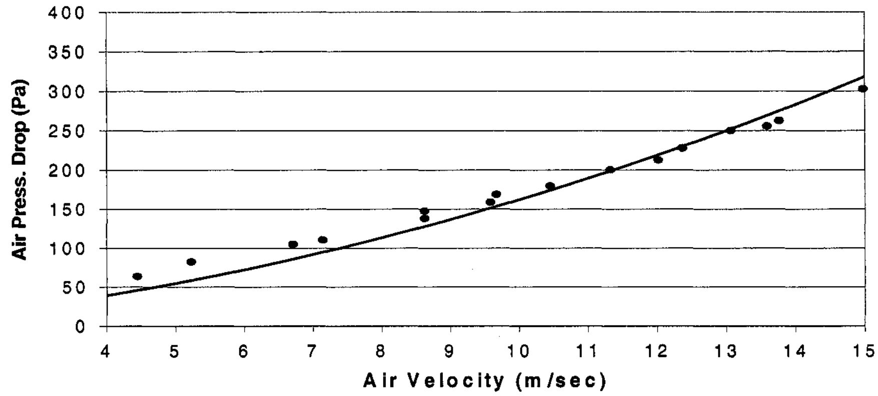

Figure 3a: Pressure drop of the air across the heat exchanger as a function of velocity (mass flow rate) at room temperature

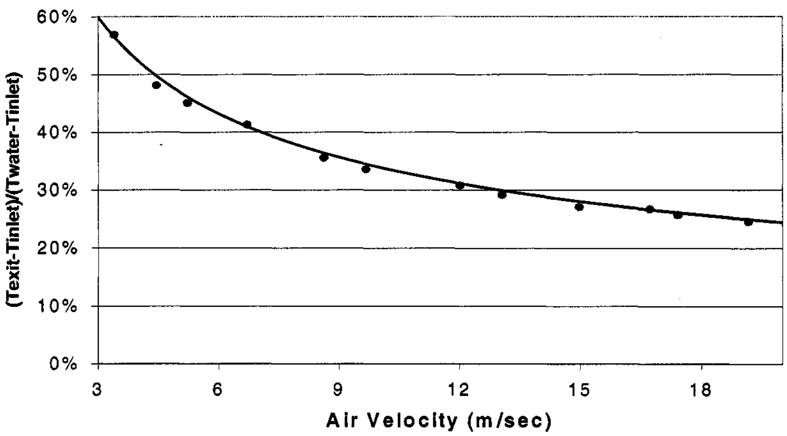

Figure 3b: Percentage of the maximum possible heat transfer

The overall heat transfer performance of the cross flow heat exchangers (both polymer and metal) are shown in Figure 4. The results are normalized with respect to frontal area and the difference in temperature of the two fluids entering the heat exchanger. Finally, Table 1 shows the performance of these micro heat exchangers versus conventional scale, maximized-performance cross flow heat exchangers, holding the pressure drop of the air across the heat exchangers constant. It can be seen that the conventional scale heat exchanger performs better than both the metal and polymer heat exchangers on a unit frontal area basis, while the micro heat exchangers have better heat transfer/weight and heat transfer/volume ratios. Also shown in Table 1 is a *predicted* performance of a micro heat exchanger, with a design similar to the polymer heat exchanger but fabricated from a metal or conductive ceramic. In this case, the design was aggressive (dimensions challenge the existing manufacturing tolerances of the authors). The predicted result is a heat exchanger that outperforms the conventional scale heat exchanger in all performance categories.

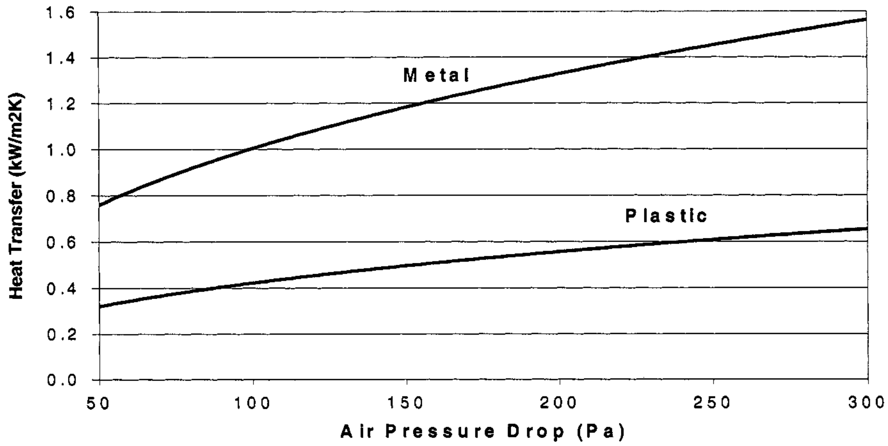

Figure 4: Heat transfer of metal and polymer micro cross flow heat exchangers

Table 1. Heat transfer performance comparison between conventional-scale and micro-scale cross flow heat exchangers.

Heat Exchanger (HE)	Δp_{air} (Pa)	$Q/(A\Delta T)$ (kW/m^2-K)	$Q/(V\Delta T)$ (kW/m^3-K)	$Q/(m\Delta T)$ (W/kg-K)
Polymer Micro HE	175	0.58	400	692
Electroless Ni Micro HE	175	1.30	1036	440
Al Micro HE (predicted)	175	3.64	910	1720
Conventional-scale HE	179	3.12	123	283

2. Heat exchanger for gas turbine component cooling

A major component in the development of advanced gas turbine engines is the increase of turbine inlet temperatures. Associated with this drive for higher turbine inlet temperatures is the need for more effective blade cooling strategies. There is, therefore, a significant effort ongoing in the gas turbine industry, federal laboratories, and academia directed at improving blade-cooling technology. Current cooling technology relies primarily on a combination of internal cooling through serpentine ribbed-coolant passages that are integrally cast in the blades or film cooling where a coolant jet is injected through a series of coolant holes on the blade surfaces. Due to the need for improved blade cooling, researchers are focusing attention on optimizing every aspect of the current technology. This includes depositing a protective coating on the blade surface, optimizing the shape and orientation of the rib-turbulators in the coolant passages, exploring film-coolant hole-shapes and coolant injection angles, optimizing the location of the coolant holes, etc. The advances in this field have been encouraging and have led to, for example, thermal barrier coatings (TBC) on blade surfaces, the use of shaped holes for film cooling and internal-rib-turbulators with V-orientations.

A new method to fabricate a micro heat exchanger directly on a metal surface is described in this paper that possibly provides another opportunity to improve blade cooling technology. The fabrication procedure begins by wrapping a LIGA-fabricated polymer template over the surface of interest[11]. A moderately dense array of microstructures (nickel or nickel-alloy) are electroplated through the template. Once the holes in the template are filled with metal, continued overplating eventually results in the overplated regions merging, producing a continuous surface that follows the contours of the substrate. For the case of turbine blade cooling, the upper nickel-alloy shroud is a few hundred micrometers thick, and the gap separating the shroud and the internal core of the turbine blade ranges between 100-1000 μm. The diameter of the microstructures can range from fifty to a few hundred micrometers, and the spacing between the microstructures is variable. To quantify the performance of such a micro heat exchanger in a gas turbine blade scenario, a schematic of a micro heat exchanger electrodeposited on a one half inch diameter tube is shown in

Figure 5. The tube will be inserted into a hot gas stream that simulates combustion gases flowing over the exterior of the blade. Coolant flows through the tube, and enters the annulus between the tube and the overplated layer through a series of coolant holes near the leading edge. The coolant passes through a field of pin fins that enhance heat transfer between the coolant and the overplated shroud. Finally, the coolant exits near the trailing edge of the tube. SEMs of the actual heat exchanger are shown in Figure 6a and 6b. The pins are 200 micrometers in diameter, spaced in a square array 1000 micrometers apart. The height of the pins is 500 micrometers.

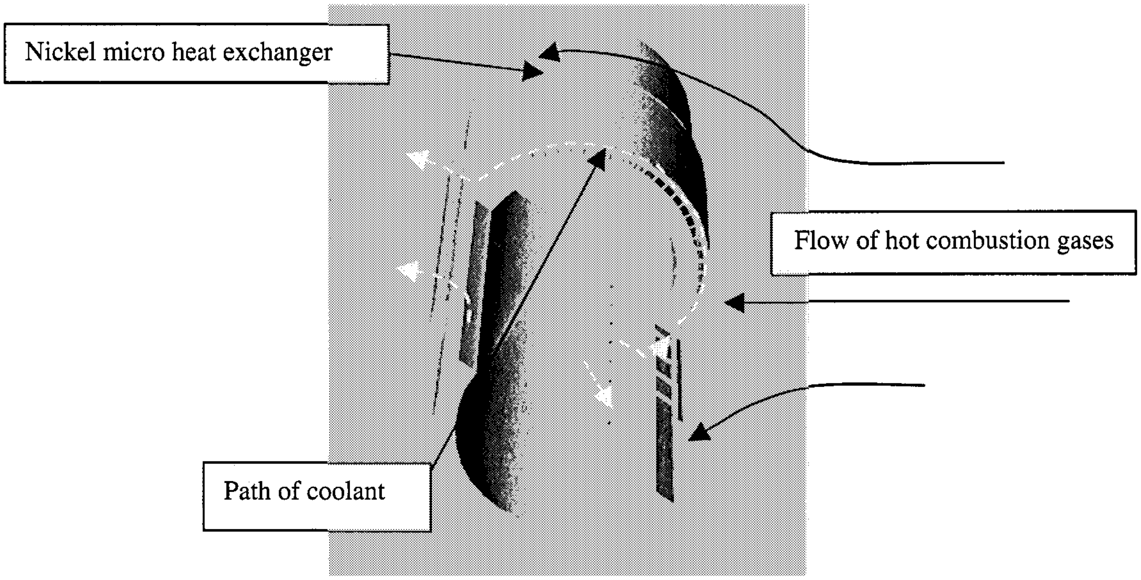

Figure 5: Schematic of micro heat exchanger on a tube

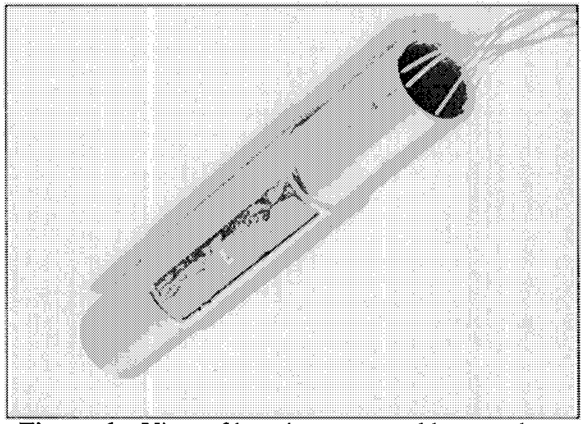

Figure 6a: View of heat instrumented heat exchanger from trailing edge.

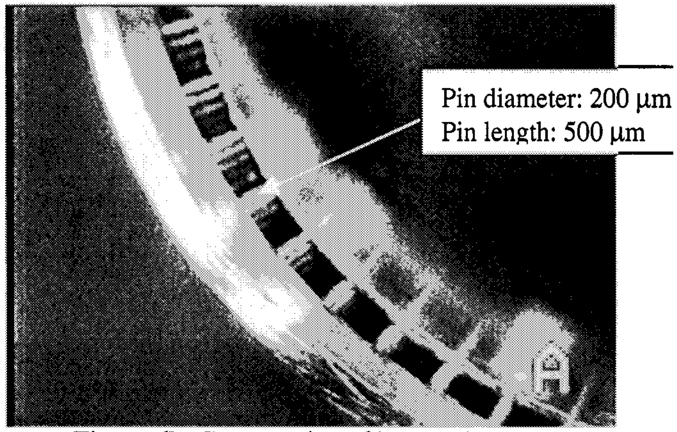

Figure 6b: Cross section of heat exchanger

Performance tests on both planar and cylindrical micro pin fin heat exchangers are in progress and should be completed by December 2001. In particular, the effective friction factor (needed for pressure drop calculations) and the effective Nussult number (needed for heat transfer calculations) will be determined as a function of coolant mass flow rates. For the case of a turbine blade, the eventual goal will be quantify the cooling effectiveness, ε, of the micro heat exchanger as a function of cooling flow rate. The cooling effectiveness is defined below (Equation 4):

$$\varepsilon = \frac{T_{gas} - T_{\text{shroud}}}{T_{\text{gas}} - T_{\text{coolant}}} \qquad \text{Equation (4)}$$

T_{gas} is characteristic temperature of the gases/environment with which the heat exchanger is exchanging energy. T_{shroud} is the temperature of the overplated (outer) surface of the heat exchanger. $T_{coolant}$ is the temperature of the coolant when it is injected into the heat exchanger. The value of ε varies between 0 and 1 depending upon the value of T_{shroud}. An effective cooling scheme provides a relatively high value of ε with a relatively low coolant mass flow rate. The typical range of ε in advanced gas turbine blades is 0.3-0.5. Improving significantly upon these numbers implies an increased cooling capability, and the potential to increase the allowable turbine inlet gas temperature and pressure.

3. Mechanical Seal

Mechanical seals are subjected to both thermal and load-induced stresses that can distort mating surfaces during operation, leading eventually to unacceptable performance (leakage, shutdown). In an effort to remove the thermal component from the list of causes leading to failure, a mechanical seal prototype has been fabricated with a *micro* pin-fin array heat exchanger located just below the load-bearing surface using a derivative of the LIGA micromachining process. The combination of pin fin geometry and micro scale makes it possible to remove friction-generated heat from the load-bearing surface extremely efficiently, thus alleviating temperature related failures with no appreciable increase in component weight or volume.

Figures 7a and 7b illustrate the coolant flow path through the prototype. Coolant (air) enters at point ① through three inlet ports located 120° apart on the shrink fit sleeve and flows from the coolant distribution annulus axially through the coolant distribution holes and into the micro heat sink at point ②. Flow continues from the outer radius of the heat sink, through the pin array and exits to atmospheric conditions at point ③. In the prototype testing, the coolant was discharged where the shaft of the pump would be located. More recent designs provide a method to route the coolant both into and away from the heat exchanger.

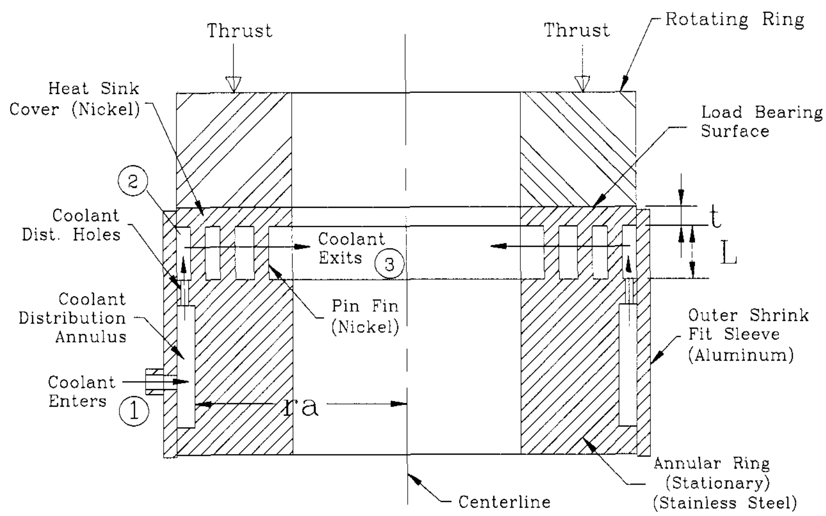

Figure 7a: Cross section elevation of prototype

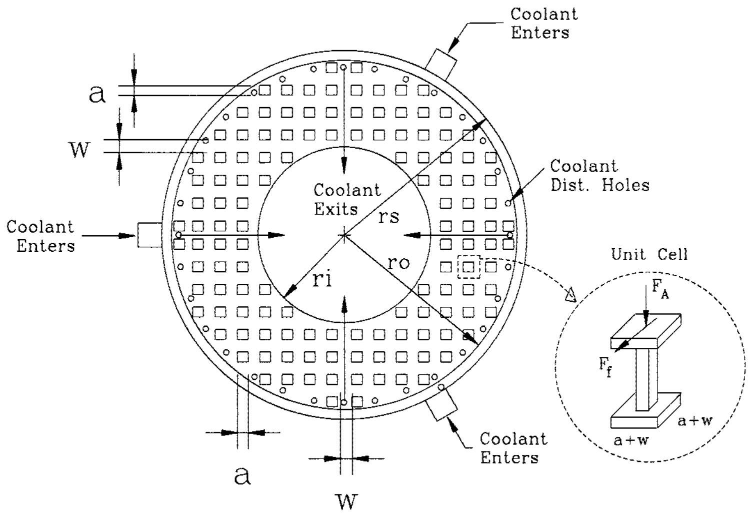

Figure 7b: Plan view of cross section of prototype

Two prototypes with different pin fin spacing and geometry were fabricated and tested. The fabrication process involved electroplating a micro heat exchanger and overplated load bearing cap on the end of a thrust washer, using a special jig[6,9]. An aluminum sleeve is shrunk fit onto the outer diameter of the stationary thrust ring, producing the prototype shown in Figure 8a. A magnified view is shown of the electroplated pin-fin array beneath the load-bearing cover is shown in Figure 8b.

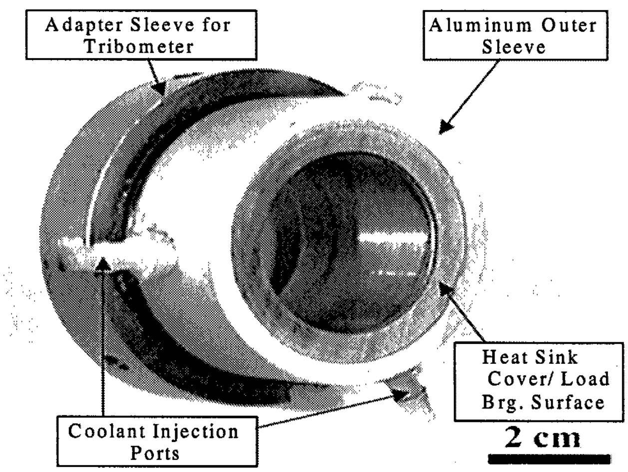

Figure 8a: Prototype Mechanical Seal

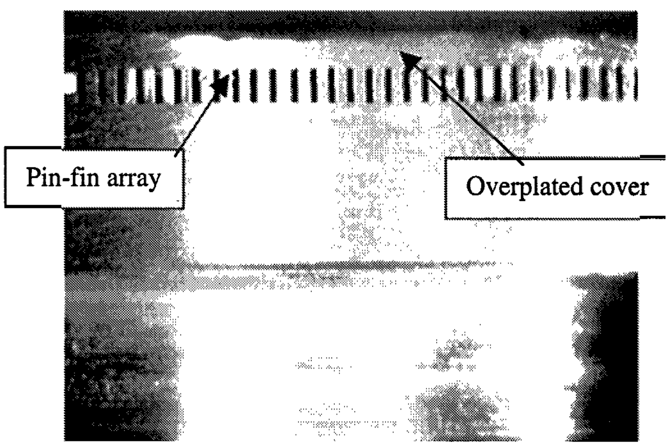

Figure 8b: Heat exchanger beneath overplated load-bearing surface

Static heat transfer experiments have been performed to quantify the performance of the micro heat exchanger using air as coolant. These experimental results compare reasonably well to relatively simple analytic models that were developed to predict performance. Once the static heat transfer tests were completed, the prototype was installed in a radial thrust washer tribometer and the temperature at the rotating/stationary ring interface was measured as a function of load, RPM and coolant flow rate through the micro heat exchanger. Typical results, shown in Figure 9, demonstrate how the temperature of the interface returns to a temperature just above ambient when the coolant is introduced. The top two curves of Figure 9 show the interface thermal time history of the actual prototype and a solid ring for the uncooled cases. The lower two curves show the thermal time history of the interface after coolant is introduced (at two different flow rates).

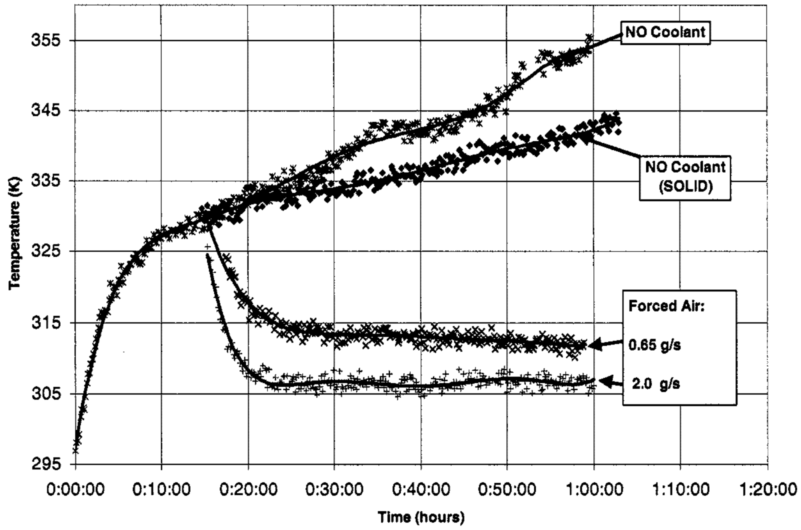

Figure 9: Prototype interface temperature as a function of coolant flow rate

The impact of this reduction in temperature is significant with regard to mechanical seal reliability. Figure 10a below shows the temperature distribution of a typical mechanical seal with the sealed fluid providing cooling at 70 °F. This result was produced using the finite element package Ansys, and taken from reference[12]. Figure 10b shows the same mechanical seal with the addition of the micro heat exchanger with a coolant supplied at 70 °F. The most outstanding result is that the micro heat exchanger maintains the maximum seal temperature at 81 °F, as compared to 145 °F for the typical mechanical seal. This result is critical to light hydrocarbon sealing applications, where a small addition of heat to the sealed fluid makes it change phase to a gas, thus creating several reliability problems including cavitation pitting and gross gassing up of the pump. Secondly, the tremendous reduction in seal temperature indicates that the dry running capability of the mechanical seal (which corresponds to zero leakage) is significantly enhanced with the micro heat exchanger. Sealing capability is further enhanced using the micro heat exchanger by a more uniform temperature distribution in both the radial and the axial directions. This results in less distortion of the seal rings and a reduction in thermal stresses that may lead to thermal cracking failures.

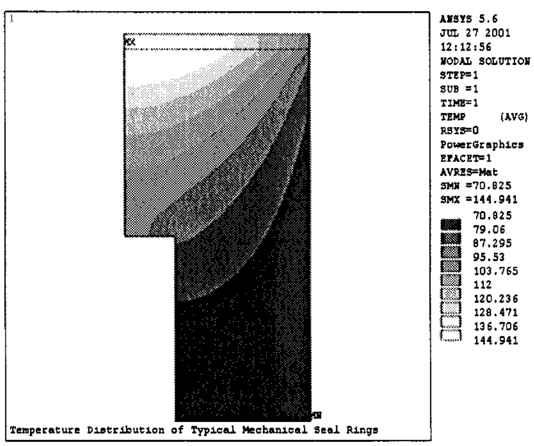

Figure 10a: Temperature Distribution of Typical Mechanical Seal (degF)

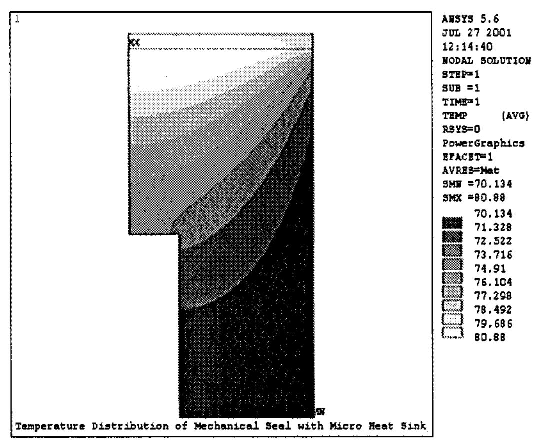

Figure 10b: Temperature Distribution of Mechanical Seal with Micro Heat Sink (degF)

4. Silicon Nitride HARMs

The advantages of micro scale with regards to heat transfer have motivated the effort to fabricate a variety of heat exchangers that have been previously discussed. Developing techniques to fabricating these heat exchangers from a wide variety of materials (alloys, a wide range of polymers, and ceramics) is one area of focus. Within the last six months, great progress has been made fabricating silicon nitride HARMS using polymer "lost molds". The ability to fabricate silicon nitride heat exchangers for all of the previously mentioned applications has great potential due to the excellent combination of stiffness, strength, thermal conductivity and wear properties of silicon nitride.

To fabricate silicon nitride parts, PMMA features such as shown in Figure 11a are molded. These features are then bonded to another PMMA part, producing an enclosed PMMA "lost mold". A silicon nitride precursor is injected into the mold, and after solidification at a relatively low temperature, the polymer mold is dissolved. The remaining solid is then sintered and the result is a fully-dense silicon nitride part such as shown in Figure 11b. Efforts are now underway to fabricate silicon nitride mechanical seals, with integral micro heat exchangers, and silicon nitride components for gas turbine engines.

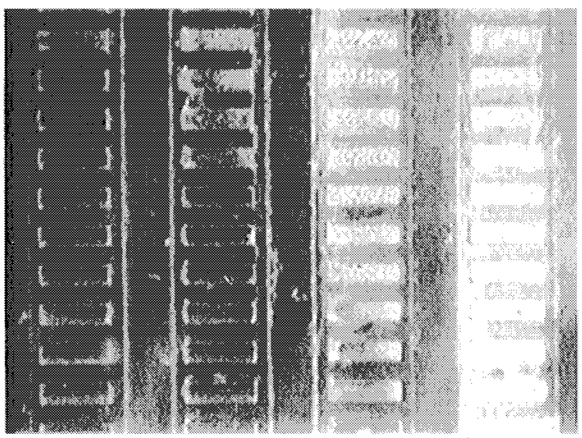

Figure 11a: Polymer "lost mold"

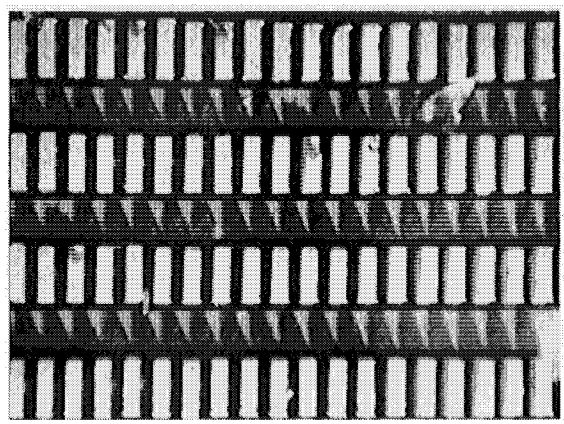

Figure 11b: Fully dense sintered silicon nitride posts

CONCLUSIONS

A number of micro heat exchangers have been fabricated for a wide range of applications. In the case of the cross flow heat exchanger, the performance the merits of heat transfer/mass or heat transfer/volume is superior to existing conventional scale cross flow heat exchangers. A micro heat exchanger can prevent the temperature of the load-bearing surface of a mechanical seal from varying, which could prevent thermal distortion related problems. The potential to fabricate a mechanical seal with integral heat exchanger from silicon nitride is an area of ongoing research. With regards to gas turbine component cooling, the feasibility of fabricating complex micro heat exchangers on nonplanar surfaces has been demonstrated. Test results, which will soon be completed, will quantify the potential value of such heat exchangers. Should the heat transfer results be favorable, the emphasis will shift from heat transfer to fabricating micro heat exchangers from appropriate materials such as high temperature, high strength nickel alloys or ceramics.

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